SYSTEMS ANALYSES OF ADVANCED BRAYTON CYCLES

FOR

HIGH EFFICIENCY ZERO EMISSION PLANTS

TOPICAL REPORT

<u>TASK 1.4.1: SCREENING ANALYSIS OF ADVANCED BRAYTON CYCLES</u>

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EXECUTIVE SUMMARY

The ultimate goal of this program is to identify the power block cycle conditions and / or configurations which could increase the overall thermal efficiency of the Baseline IGCC by about 8% on a relative basis (i.e., 8% on a heat rate basis). This document presents the cycle conditions and / or the configurations for evaluation in an initial screening analysis. These cycle conditions and / or configurations for investigation in the screening analysis are identified by literature searches and brain storming sessions. The screening analysis in turn narrows down the number of promising cases for detailed analysis.

APPROACH

Simulations of the power blocks (identified by the literature searches and brainstorming sessions of having a potential for increasing the thermal efficiency of the Baseline Case significantly) are performed on Thermoflex. The syngas composition as established in the Baseline Case is used in these simulations. The steam/BFW interchanges between the power block and the syngas generation (gasification) plant are taken into account in the bottoming cycle. The flow rates of the steam/BFW streams are adjusted in proportion to the fuel consumption of the power block. The net thermal efficiency of the overall plant is estimated by accounting for the power required both by the power block and by the gasification plant. Based on these results, cycle conditions and / or configurations are then be proposed for detailed analysis in the next step of this program that have a potential for significant improvement in the overall plant thermal efficiency (by about 8%) over the Baseline Case.

Selection of Cases for Detailed Analysis

The following lists the proposed criteria for selecting the cycle conditions and / or configurations evaluated by the screening analysis for the detailed analysis of this program:

- Simplicity of configuration and controllability
- High overall IGCC plant thermal efficiency
- Minimum increase in pressure ratio over the Baseline Case while reaching the thermal efficiency goal
- Potential for lowering NOx

Cycle improvements or combinations of two or more of the improvements evaluated in the screening analysis are then selected for detailed analysis as described in the following section. Some examples of combinations of cycle improvements may be:

- Gas turbine compressor intercooling with turbine reheat and chemical recuperation.
- Gas turbine inlet air fogging with turbine reheat and chemical recuperation.

Cycles Proposed for Screening Analysis

The following describes the cycle conditions and / or configurations identified for the power block by the literature searches and brainstorming sessions of having a potential for increasing the thermal efficiency of the Baseline Case.

Increased Firing Temperature / Blade Metal Temperature

The effect of raising the firing temperature of the Baseline Case gas turbine is quantified for a given metal temperature of the 1st stage stator blades. Pressure ratio is varied to obtain the maximum plant thermal efficiency. A map of firing temperature (at the optimum pressure ratio) versus cycle efficiency is generated while adjusting the blade metal / TBC temperatures such that the coolant amounts to each set of blades remain at the same values as the Baseline Case gas turbine. This map is superimposed on to Figure A1.4.1- 1 (which shows projected increases in blade metal / TBC temperatures as increases in the firing temperature are realized in the future) to check for reasonableness of the blade metal / TBC temperatures used in this analysis. The minimum firing temperature along with the corresponding blade metal / TBC temperatures are then selected for use in the remainder of this screening analysis task with the goal of achieving the efficiency target of this program.

Pressure Gain Combustor

A pressure gain combustor produces an end-state stagnation pressure that is greater than the initial state stagnation pressure. An example of such a system is the constant volume combustion in an ideal spark ignited engine. Such systems produce a greater available energy in the end state than constant pressure systems. It was shown by Gemmen, Richards and Janus [1994] that the heat rate of a simple cycle gas turbine with a pressure ratio of 10 and a turbine inlet temperature of ~1200ºC (2200ºF) could be decreased by more than 10% utilizing such a constant volume combustion system. Pulse combustion which relies on the inherent unsteadiness of resonant chambers can be utilized as a pressure gain combustor. Research continues at the U.S. DOE and at NASA for the development of pressure gain combustors.

The impact on the plant thermal efficiency by utilizing a pressure gain combustor in the gas turbine is quantified.

High Efficiency Exhaust Diffuser

Meruit Inc. [Fonda-Bonardi, 1996] has developed an Annular Recirculating Diffuser concept which is expected to improve the efficiency of a gas turbine engine by 3% by reducing the exhaust loss in the turbine section. In cycles employing exhaust heat recovery such as in combined cycle applications, the net overall cycle efficiency gain is expected to be lower however. The impact on the overall IGCC plant efficiency is quantified by incorporating this type of diffuser.

Inlet Air Fogging

Roughly 50% of the power developed by the turbine in a gas turbine is used in its compressor. An approach to reducing this large parasitic load of air compression in a gas turbine is to introduce liquid water into the suction air [Utamara et. al., 1999; Bhargava and Meher-Homji, 2002]. The water droplets will have to be extremely small in size and be in the form of a fog to avoid impingement on the blades of the compressor causing erosion. As the water evaporates within the compressor from the heat of compression, the air being compressed is cooled which in turn causes a reduction in the compressor work. Note that the compression work is directly proportional to the absolute temperature of the fluid being compressed.

A benefit in addition to increasing the specific power output of the engine is the reduction in the NOx due to the presence of the additional water vapor in the combustion air. A number of gas turbines have been equipped with such a fogging system. Care should be taken, however, in specifying the water treatment equipment since high quality demineralized water is required as well as in the design of the fogging system to avoid impingement of the compressor blades with water droplets.

The impact on the plant thermal efficiency by the addition of gas turbine inlet fogging is quantified.

Inverse Cycle

The "inverse cycle" proposed by many investigators in the past (http://www.energytech.at/kwk/portrait_kapitel-2_6.html#h4) consists of reducing the back pressure on the gas turbine exhaust to sub-atmospheric pressure and utilizing a blower installed downstream of the HRSG to pressurize the flue gas to atmospheric pressure so that it may be discharged to the atmosphere. A cooler installed between the HRSG and the blower helps reduce the parasitic blower power consumption. Such a cycle has been touted for applications where a low calorific value fuel gas containing a significant fraction of hydrogen is available at a low pressure. In such cases, the gas turbine pressure ratio may be increased utilizing the flue gas blower to reduce the gas turbine exhaust pressure, instead of by increasing the turbine inlet pressure and having to compress the large volume of the low calorific value fuel gas to the correspondingly higher pressure required by the gas turbine combustor.

Reheat Gas Turbine

Gas turbine efficiency may be improved by incorporation of a reheat or sequential combustor. Figure A1.4.1-2 depicts the reheat gas turbine cycle. The following lists the main features of this cycle:

- Increased Cycle Efficiency
  - Alstom's Approach while Maintaining Lower Firing Temperature
  - Approximately 2% Improvement in Combined Cycle Heat Rate
  - Other Gas Turbine Vendors Evaluating this Option
- Reduced NOx Emissions
  - Due to Lower Firing Temperature
  - NOx Destruction in Reheat (Sequential) Combustor

Because of the above listed advantages, evaluation of the reheat cycle is included in this screening study. The impact on the plant thermal efficiency by the addition of a reheater in the gas turbine is quantified. Included in this analysis is the optimum placement of the reheat combustor, i.e., the optimum pressure ratio of the high pressure turbine providing the vitiated air to the reheat combustor.

Intercooled and Reheat Gas Turbine

Another approach to reducing the parasitic load of air compression in a gas turbine is to incorporate intercooling. Intercooling is justified from an overall cycle thermal efficiency standpoint however at very high pressure ratios. Since the Advanced Brayton cycle with the high firing temperature in combination with reheat is expected to optimize at very high pressure ratio, intercooling of the compressor is included in this screening study. Figure A1.4.1-3 depicts the intercooled / reheat gas turbine cycle.

The impact on the plant thermal efficiency by the addition of an intercooler in the reheat gas turbine compressor is quantified. Included in this analysis is the optimum placement of the intercooler, i.e., the optimum pressure ratio of the low pressure compressor providing the air to the intercooler.

Supercritical Rankine Bottoming Cycle

The bottoming cycle used by GE for the H technology gas turbine based combined cycles consists of subcritical conditions. The bottom cycle as configured by UCIrvine utilizing literature data published by GE consists of a triple pressure superheat-reheat cycle with steam conditions at the throttle of the high pressure steam turbine of 165 bar / 566ºC or 2400 psig / 1050ºF and those of the reheated steam at the inlet of the steam turbine of 24 bar / 566ºC or 345 psig / 1050ºF. Use of supercritical steam cycle conditions in a high firing temperature gas turbine (with a correspondingly high exhaust temperature) may have a potential of increasing the overall combined cycle thermal efficiency significantly. Figure A1.4.1-4 presents the thermal efficiency of the steam Rankine cycle for various subcritical and supercritical conditions [Kitto, 1996]. A current State-of-the-Art steam cycle consists of 290 bar / 580ºC / 600ºC or 4200 psi /

1080ºF / 1110ºF while the European Thermie Project is scheduled to demonstrate in the year 2008, cycle conditions of 375 bar / 700ºC or 5439 psi / 1292ºF and the projected thermal efficiency (HHV) of > 45%. Table A1.4.1 - 1 summarizes some of the supercritical steam conditions being offered currently or being developed [Armstrong, Abe, Sasaki and Matsuda J., 2003; Ashmore, 2006; Kjaer (Elsam Engineering A/S); Retzlaff and Ruegger, 1996; Torre, 2003].

Table A1.4.1 - 1: Supercritical Steam Cycles

Manufacturer/Study	Steam Conditions	Reheat
Hitachi	248 barg / 600 ^{o}C / 610 ^{o}C (3600 psig / 1112^{o}F / 1130 ^{o}F)	Single
Siemens	300 bar /600^{o}C / 620^{o}C (4350 psi / 1112^{o}F / 1148^{o}F)	Single
GE (1980s EPRI)	2482 bar / 593^{o}C / 593C (4500 psi / 1100^{o}F / 1100 ^{o}F)	Single
GE Philo 6 Plant	2482 bar / 621^{o}C (4500 psi/1150 ^{o}F)	
THERMIE Program (Study)	375 bar / 700 ^{o}C (5439 psi / 1292 ^{o}F) 1st Reheat: 120 bar/720 ^{o}C (1740 psi/1328 ^{o}F) 2nd Reheat: 23.5 bar/720 ^{o}C (340 psi/1328 ^{o}F)	Double

Chemical Recuperation

It may be possible to recover a portion of the high temperature heat available in the gas turbine exhaust to endothermally react the H_2 rich decarbonized syngas with the residual amounts of CO_2 also present in the syngas by the following “reverse shift” reaction:

$$H_2 + CO_2 = H_2O + CO$$

It is expected that the reaction will move in the reverse shift direction since the concentration of the H_2 in the decarbonized syngas is very high while that of the CO is very low.

The impact on the plant thermal efficiency by the addition of chemical recuperation by which exhaust heat from the gas turbine is recycled to its combustor is quantified.

Humid Air Cooling of Gas Turbine Blades

The advantages with steam cooling of the gas turbine blades over air cooling are:

- Gas turbine compression power is reduced
- Thermal dilution losses in the turbine are minimized when closed circuit cooling is utilized
- Momentum losses in the turbine are minimized again when closed circuit cooling is utilized.

- NOx emissions are reduced since the gas turbine combustor exit temperature is reduced for a given rotor inlet temperature.

A disadvantage of utilizing closed circuit steam cooling however, is that heat absorbed by the steam within the turbine enters the bottoming (steam) cycle by passes the topping (gas turbine) cycle. With open circuit air cooling of the turbine blades, the bypassing of the heat is avoided but this method of cooling does not have the above advantages listed for closed circuit steam cooling. Humidification of the cooling air utilizing low temperature heat has the potential of reducing the major penalty associated with air cooling which is the increase in the parasitic air compression power requirement.

The impact on the plant thermal efficiency by utilizing an air cooled gas turbine with humidification of the cooling air utilized in the high pressure stages of the turbine is quantified. An SCR is included to reduce the NOx emissions since NOx emissions from the gas turbine would be higher due to the higher operating temperature of the combustor of this non-steam cooled gas turbine.

HAT – Combined Cycle

A potential exists to synergistically combine the HAT cycle with the combined cycle to improve the overall thermal efficiency of an integrated gasification power plant. Figure A1.4.1-5 depicts the proposed cycle. The high pressure superheated steam generated in the gasification section of the plant is utilized in a back pressure steam turbine. The heat from the exhaust steam is recovered by condensing it in a high pressure condenser against HAT humidifier circulating water. The gas turbine consists of humid air cooling of the turbine blades rather than steam cooling since it is expected that the power block will be started up on natural gas without the gasification plant on-line which is the only source for the steam. The resulting overall plant thermal efficiency is quantified.

Liquid Water-Cooled HAT

Cooling of the turbine blades with liquid water has been proposed in the past and a detailed theoretical analysis was performed by the National Advisory Committee for Aeronautics [Byron and Livingood, 1947]. Since the HAT cycle cannot take advantage of steam cooling, water cooling with the subsequent use of the hot water exiting the turbine (after performing the blade cooling function) in the humidifier of the HAT cycle has the potential of raising the overall cycle thermal efficiency. The performance of the "HAT-Combined Cycle" may be improved by this liquid water cooling method.

Oxy Combustion Gas Turbine

Various cycles have been proposed where O_2 rather than air is utilized for the combustion of the fuel. Examples of such cycles are the (1) Graz cycle, (2) Partial Oxidation cycle (which resulted from study of fundamental Brayton cycle principles as put forth by Northwestern and improved upon by Gas Technology Institute), and (3) Clean Energy Systems cycle. A single oxy combustion cycle will be selected by this screening analysis task for the later detailed analysis.

Vortex Combustion Gas Turbine

The Trapped Vortex Combustor (TVC) has the potential for numerous operational advantages over current gas turbine engine combustors. These include lower weight, lower pollutant emissions, effective flame stabilization, high combustion efficiency, and operation in the lean burn modes of combustion. The TVC concept grew out of fundamental studies of flame stabilization and is a radical departure in combustor design using swirl cups to stabilize the flame. Swirl stabilized combustors have somewhat limited combustion stability and can blow out under certain operating conditions. On the other hand, the TVC maintains a high degree of flame stability because the vortex trapped in a cavity provides a stable recirculation zone that is protected from the main flow in the combustor. The second part of a TVC is a bluff body dome which distributes and mixes the hot products from the cavity with the main air flow. Fuel and air are injected into the cavity in a way that it reinforces the vortex that is naturally formed within it.

The TVC may be considered a staged combustor with two pilot zones and a single main zone, the pilot zones being formed by cavities incorporated into the liners of the combustor [Burrus et. al., 2001]. The cavities operate at low power as rich pilot flame zones achieving low CO and unburned hydrocarbon emissions, as well as providing good ignition and the lean blowout margins. At higher power conditions (above 30% power) the additional required fuel is staged from the cavities into the main stream while the cavities are operated at below stoichiometric conditions. Experiments have demonstrated an operating range that is 40% wider than conventional combustors with combustion efficiencies of 99%+. Use of the TVC combustor holds special promise as an alternate option for suppressing the NOx emissions in syngas applications where pre-mixed burners may not be employed. Organizations actively involved in the development of such combustors include General Electric and Ramgen. A semi-quatitative analysis will be made of the use of the TVC in an IGCC to assess if it has a significant impact on the plant thermal efficiency.

RESULTS AND DISCUSSION

Gas Turbine Cycle Configuration

Pressure Gain Combustor

Figure A1.4.1-6 shows the fuel (syngas) to air ratio plotted against the calculated compressor discharge pressure and the combustor discharge pressure while maintaining the same combustor exhaust temperature as that in the Baseline Case of 1433°C or 2611°F. A 4% pressure loss was also assumed as in the Baseline Case. Note that the compressor discharge temperature changes as its discharge pressure changes. As can be seen from the data presented in the plots, the pressure gain expressed as the ratio of the combustor discharge pressure to the compressor discharge pressure varies by as much as 2.3 to 3.0 as the compressor discharge pressure is varied from 5 bar to 20 bar.

The complete gas turbine cycle along with the steam bottoming cycle ware next simulated for a compressor discharge pressure of 8.74 bar which provides a turbine inlet pressure same as in the Baseline Case gas turbine of 23.5 bar. The resulting net heat rate of the IGCC plant was significantly reduced, by as much as 7%. Next a sensitivity case was simulated to assess the impact on the overall IGCC plant heat rate if only half of this pressure gain could be actually realized due to much higher losses. The required compressor discharge pressure had to be increased to 17.4 bar to obtain the same turbine inlet pressure of 23.5 bar. The resulting net heat rate of the IGCC plant was still significantly impacted, reduced by almost 3%. Thus, the pressure gain combustor has the potential to make a significant positive impact on the IGCC plant performance. Major challenges exist, however, with respect to interfacing the pressure gain combustor which tends to be cyclic in operation with the gas turbine compressor and turbine which require steady flows.

High Efficiency Exhaust Diffuser

According to Meruit Inc., the gas turbine exhaust diffuser can be designed to have a coefficient of performance (Cp) as high as 0.9 utilizing their proprietary design. As a reference point, the Cp of a “conventional” diffuser is typically about 0.6. Before a detailed analysis of the diffuser is conducted in order to verify Meruit’s claim, a sensitivity case was developed to establish the upper limit for efficiency gain utilizing a diffuser approaching a Cp of 1.0. The results of this analysis indicated that the power output of the gas turbine is increased by as much as 3.5% but at the expense of a reduction of as much as 15°C or 27°F in the exhaust temperature. The steam turbine power output is consequently reduced with an overall combined cycle heat rate improvement of about 1% over the Baseline Case.

Inlet Air Fogging

Using 0.5% overspray (expressed as % of saturated air flow) which is typically the maximum amount beyond which the gas turbine warranties do not hold, the overall plant performance is actually poorer. The compressor discharge temperature is reduced from 487°C or 908°F (for the Baseline Case) to 447°C or 836°F indicating a significant reduction in compression power but on the other hand, the fuel to air ratio is increased by 3.7% over the Baseline Case negating the savings in compression power. Secondary effects causing a further reduction in the efficiency of this Inlet Fogging Case are: (1) due to the higher moisture content of the air entering the combustor or the gas entering the turbine, a reduction the firing temperature from 1392°C or 2538°F (for the Baseline Case) to 1388°C or 2530°F to maintain the same 1st stage stator blade temperatures, (2) due to the lower compressor discharge temperature for the Inlet Fogging Case, a reduction from 4,747 kW generated by the extraction expander to 4,559 kW and also (3) a reduction of 9.5 GJ/hr or 9 MMBtu/hr of heat available for LP steam generation downstream of this expander. The net reduction in power generated by this Inlet Fogging Case over the Baseline Case is 1,361 kW or 0.36%.

Inverse Cycle

A simulation was performed to quantify the performance improvement, if any, of the Baseline Case when equipped with a blower to draw a vacuum in the gas turbine exhaust such that HRSG exhaust. The exhaust pressure was reduced to 0.68 bar or 9.9 psia versus the 1.014 bar or 14.7 psia for the Baseline Case. The HRSG exhaust was first cooled to 27°C or 80°F using cooling water followed by the blower to compress the gas back up to the ISO atmospheric pressure. The net heat rate actually increased by almost 2% even after making the following optimistic assumptions: (1) 90% blower polytropic efficiency and (2) less than 7.6 cm or 3 in WC pressure drop for the flue gas cooler.

The Inverse Cycle may be useful in gas turbine based cycles where the flue gas contains a large fraction of water vapor (such as “wet cycles”). In such applications, the quantity of gas to be compressed in the blower would be much smaller than the working fluid within the gas turbine when the flue gas is cooled below its dew point upstream of the blower such that a significant fraction of the water vapor is condensed out.

Reheat Gas Turbine

A reheat combustor is installed between the 1st and 2nd stages of the turbine. The pressure ratio is increased to 36 which is the highest for a commercially offered non-intercooled land-based gas turbine (Rolls Royce Trent 60 WLE, an aero engine) at ISO conditions. The reheat combustor outlet temperature is reduced in order to limit the temperature of the gas leaving the last stage to around 650°C or 1200°F (actual temperature obtained is 669°C or 1237°F) such that strength in the roots of the long and uncooled last stage blades is maintained. Furthermore, use of advanced superheat and reheat steam temperatures of 621°C or 1150°F for the bottoming cycle is facilitated without having very large temperature differences between the gas turbine exhaust and the steam such that the irreversibility in heat transfer is similar to that in the Baseline Case. The resulting reduced rotor inlet temperature for the 2nd stage turbine is 1345°C or 2453°F while the 1st stage rotor inlet temperature is kept close to that of the Baseline Case (1391°C or 2536°F versus 1392°C or 2538°F for the Baseline Case). The net increase in power generated by the plant over the Baseline Case (on a constant coal consumption basis) is significant, about 9 to 10 MW or more than 2%.

Intercooled and Reheat Gas Turbine

An intercooler is installed in the compressor which cools the air leaving the LP compressor (having a pressure ratio of 2.9) against cooling water to a temperature of 27°C or 80°F. A higher overall pressure ratio may be realized with intercooling without letting the compressor discharge temperate becoming excessive. Higher pressure ratio in turns allows raising the rotor inlet temperature of the 2nd stage turbine to that of the Baseline Case while limiting the gas turbine exhaust temperature to around 650°C or 1200°F (actual temperature obtained is 670°C or 1238°F) such that strength in the roots of the long and uncooled last stage blades is maintained. The reheat combustor is again installed between the 1st and 2nd stages of the turbine. The overall pressure ratio is increased to 42 which is close to that of the GE LMS100 intercooled gas turbine which has a pressure ratio of 41 at ISO conditions. Again, use of advanced superheat and reheat steam temperatures of 621°C or 1150°F for the bottoming cycle is facilitated without having very large temperature differences between the gas turbine exhaust and the steam such that the irreversibility in heat transfer is similar to that in the Baseline Case. The net increase in power generated by the gas turbine over the Reheat Case (on a constant syngas input basis) is insignificant however. At high pressure ratios intercooling may be desirable to limit the compressor discharge temperature which eases the challenges in the design of the compressor and the required materials of construction, as well as to reduce the formation of NOx within the combustor of the gas turbine.

Chemical Recuperation

The amount of heat converted to chemical energy by adding a shift reactor downstream of the Selexol unit (and prior to syngas humidification) in the Baseline Case while operating this reactor at an isothermal temperature of 510°C is 12.35 GJ/hr. The net increase in electric power for the IGCC plant is estimated to be 0.5 MW which is corresponds to only a 0.13% reduction in net heat rate. The heat reduction is too small to justify addition of the shift reactor and the associated heat exchange equipment.

Humid Air Cooling of Gas Turbine Blades

The steam cooling of the 1st stage turbine blades of the Baseline Case was replaced with humid air cooling. The required amount of compressor discharge air was cooled, humidified, preheated against the air humidifier in-coming air and then used in the 1st stage turbine stator and rotor blades. The moisture content of the humid air was 40% (mass basis). The overall system performance did not change significantly over the Baseline Case. This type of cooling may be considered for applications where a steam cooled gas turbine is not preferred such as in simple cycle applications or where close coupling of the Brayton cycle and the bottoming Rankine cycles is not desirable.

HAT – Combined Cycle

An air-cooled HAT-Combined Cycle was simulated where the low temperature heat available from within the cycle as well as that available in the gasification island was recovered for humidification of the compressed air of the HAT cycle while the higher temperature heat was recovered to generate steam and utilized in a back pressure steam turbine. The results showed that the net overall plant efficiency of this air-cooled HAT was essentially the same as the Baseline Case consisting of the steam cooled gas turbine. There is a potential to improve the overall plant heat rate by utilizing a liquid water-cooled HAT and thus, this water-cooled HAT case is recommended for the detailed analysis task. Ultra low NOx emissions are expected for this HAT case based on results of previous work.

Oxy Combustion Gas Turbine

Two types of oxy combustion cycles were screened, one consisting of the “OX Gas Turbine” which has both the high pressure and reheat combustors operating under oxidizing conditions and the other consisting of the “POX Gas Turbine” which has the high pressure combustor operating under sub- stoichiometric conditions while the reheat combustor operates under oxidizing conditions. In the OX Gas Turbine configuration shown in Figure A1.4.1-7, all of the oxygen required is sent through the first combustor and the syngas flow is split between these two combustors. In the POX Gas Turbine configuration shown in Figure A1.4.1-8, the entire syngas is supplied to the first combustor and the oxygen flow is split between the two combustors. In either case, the syngas is desulfurized but not decarbonized. These cycles involve post combustion carbon capture, the gas stream leaving the condenser of the cycle consisting mainly of CO_2. The O_2 is supplied by an LP ASU since N_2 dilution in the combustor of the gas turbine (typically utilized for NOx control and power augmentation) is not desirable since the CO_2 is captured downstream of the gas turbine.

Screening analysis of these two types of oxy combustion cycles indicated that the partial oxidation cycle has a heat rate advantage of about 4%. The advantage of the POX Gas Turbine is due to the fact that it does not require all the O_2 to be compressed to the HP combustor pressure while the syngas is available at high pressure. This would also result in a reduction in the cost of the O_2 compressor which tends to be a costly machine. An added advantage of the POX Gas Turbine is that the HP combustor operating under sub-stoichiometric conditions minimizes the formation (if any) of NO_x. On the other hand, the POX Gas Turbine requires the HP turbine in addition to the HP combustor to operate in partial oxidation mode. Potential challenges for the gas turbine are (1) due to the metallurgical issues such as H_2 embitterment and metal dusting within the partial oxidation combustor as well as the HP turbine, (2) soot formation within the partial oxidation combustor and (3) design of the high pressure turbine seals to prevent leakage of the CO and H_2 at the high operating temperature and pressure. A large addition of steam may be required to circumvent Concerns 1 and 2 while a buffer gas such as N_2 (supplied by the ASU) may be required for the seals (Concern 3). Assuming that these technical hurdles can be overcome, the POX with its significant performance advantage is worthy of more detailed analysis.

Vortex Combustion Gas Turbine

A potential advantage for the vortex combustor in an IGCC application is that it can do away with the requirement for thermal diluent addition in the combustor of the gas turbine for NOx control while achieving ultra low NOx emissions. As explained in the following, however, dileunt addition via syngas humidification and / or N_2 supplied by an EP ASU, in IGCC plants with pre-combustion carbon capture, has the advantage of lowering the plant heat rate. The NOx emissions are limited to 15 ppmVd (15% O_2 basis) with dileunt addition and if a lower NOx emission is required, then an SCR would have to be installed in within the HRSG. The SCR can reduce the NOx down to about 3 ppmVd (15% O_2 basis). The cost and heat rate penalty as explained under the “Low NOx Sensitivity Case” of “Task 1.3 - First Detailed Systems Study Analysis – Baseline Case” of the “Results and Discussion” section of this report, are very small. The major advantage of the vortex combustor in such pre-combustion carbon capture IGCC applications is that the NH_3 handling system of an SCR is eliminated. Thus, the evaluation of this case will be given lower priority than the other cases recommended for evaluation under this study project.

In an IGCC with a gas turbine utilizing a "diffusion" type combustors, diluent addition is required to the syngas in order to reduce the NOx generation. Two types of diluents are available in an IGCC plant, water vapor introduced into the syngas stream by direct contact of the syngas with hot water in a counter-current column while recovering low temperature waste heat and / or N_2 supplied by an elevated pressure air separation unit. The choice of the diluent depends on a number of factors such as

- amount of low temperature waste heat available for the humidification operation and
- amount of excess N_2 available from the air separation unit.

The amount of low temperature waste heat available in a gasification plant in turn depends primarily on the gasification heat recovery system employed (i.e., the extent to which cooling of the raw gasifier effluent is accomplished in a syngas cooler before the syngas is quenched / scrubbed with water). On the other hand, the amount of N_2 available as a diluent for the gas turbine depends on

- the specific O_2 consumption of the gasifier - the amount of N_2 produced by the air separation unit is lower when the specific O_2 consumption of the gasifier is lower and
- the type of gasifier feed system - dry feed systems utilize significant portions of the N_2 as lock hopper pressurization gas as well as in the drying and transport of the coal into the gasifier and only the remaining amount of N_2 is available for gas turbine injection.

In the case of the liquid slurry fed gasifier (GE type) selected for these near zero emission IGCC plants with pre-combustion carbon capture, the specific O_2 consumption tends to be high and so enough N_2 is available from the ASU for gas turbine combustor injection.

For IGCC applications, EP ASUs are preferred over LP ASUs when the oxygen and nitrogen product can be used at elevated pressures. The feed air pressure for an LP ASU is in the range of 3.5 to 6 bar (50 to 90 psig) while the feed air pressure for an EP ASU is set typically around 15 bar (200 psig). The operating pressure of the ASU distillation operation affects the bubble point of the liquid being distilled in the cold box. The higher the operating pressure, the less severe the cold box temperature is. Furthermore the cold box equipment pressure drops as a percentage of to inlet air pressure are also reduced as the cold box operating pressure is increased. The result is a reduced pressure ratio of the incoming air to that of the outgoing streams (O_2 and N_2). If the O_2 and the N_2 leaving the cold box can be utilized within the gasification plant at the product supply pressure or higher, then a net increase in the overall IGCC plant efficiency is realized. When the N_2 after further compression is introduced into the combustor of the gas turbine provides extra motive fluid for expansion in the turbine in addition to reducing the NOx emissions. Results from previous studies have indicated that about 2% reduction in both the plant heat rate and plant cost may be realized by utilizing the EP ASU over the LP ASU.

Next, for the liquid slurry fed total quench gasifier (GE type) with shifting of the syngas as in these near zero emission IGCC plants with pre-combustion carbon capture, a large amount of low temperature waste heat is generated. The low temperature waste heat can be recovered for fuel gas humidification to provide both motive fluid and thermal diluent in the gas turbine. The humidification operation consists of counter-currently contacting the syngas with hot water in a packed column to simultaneously transfer heat and mass (water vapor) into the fuel gas stream from the water stream. The evaporation of water in the presence of syngas within the column occurs at a temperature much lower than the boiling point of water. Thus, the heat required for this evaporation process may be provided by circulating the water leaving the column through the low temperature waste heat recovery exchanger located downstream of the shift unit. Thus syngas humidification allows capture of waste heat and lowers the overall IGCC plant heat rate.

Advanced Materials Technology

Increased Firing Temperature

Table A1.4.1 -2 summarizes the results of the screening analysis where the gas turbine firing temperature is increased over the Baseline Case in order to reduce the net heat rate of the IGCC. The firing temperature along with the blade metal temperatures were increased in nominal 100°C increments over those in the Baseline Case while the gas turbine pressure ratio was increased to maintain the exhaust temperature similar to that of the Baseline Case gas turbine while operating on natural gas (607°C or 1125°F). The steam bottoming cycle superheat and reheat temperatures for these higher firing temperature gas turbines were increased to the same values as those for this Baseline Case gas turbine operating on natural gas, i.e., 566°C or 1050°F (triple pressure subcritical steam cycle).

Table A1.4.1 - 2: Effect of Raising Firing Temperature on IGCC Performance

	Baseline Case	**Nominal 100°C Increase in Rotor Inlet**	**Nominal 200°C Increase in Rotor Inlet**	**Nominal 300°C Increase in Rotor Inlet**
1st Stage Rotor Inlet Temperature	1392°C (2538°F)	1502°C (2736°F)	1611°C (2932°F)	1722°C (3131°F)
Combustor Outlet Temperature	1433°C (2611°F)	1544°C (2811°F)	1655°C (3011°F)	1766°C (3211°F)
Increase in Blade Metal Temperatures over Baseline Case	-	108°C (195°F)	223°C (402°F)	338°C (608°F)
Pressure Ratio	24	30.4	44.4	63.5
Compressor Discharge Temperature	487°C (908°F)	538°C (1001°F)	630°C (1166°F)	724°C (1335°F)
Increase in Net Plant Efficiency over Baseline Case	-	3.6%	5.9%	8.0%

As can be seen, the required gas turbine firing temperature to realize an 8% decrease in the heat rate over the Baseline Case is as high as 1722°C or 3131°F (versus 1392°C or 2538°F for the Baseline Case) while the pressure ratio has to be increased to as high a value as 63.5 (versus 24 for the Baseline Case). A combination of increased firing temperature along with cycle modifications such as intercooling (based on the previous results, intercooling does not hurt the cycle performance while limiting the discharge temperature of the air) and / or reheat may be desirable in order to limit the increase in firing and blade temperatures.

Next, a case is developed to reduce the pressure ratio of the 1722°C firing temperature gas turbine while allowing the exhaust temperature increase to around the 650°C or 1200°F constraint discussed previously (actual exhaust temperature for this case is 656°C or 1212°F). The corresponding pressure ratio for this case was 49.9. The steam bottoming cycle superheat and reheat temperatures were increased to 621°C or 1150°F to take full advantage of the higher gas turbine exhaust temperature. The net IGCC plant heat rate is essentially unaffected by reducing the gas turbine pressure ratio as long as the steam superheat and reheat temperatures are increased to limit the irreversibility in heat transfer in the HRSG.

Supercritical Rankine Bottoming Cycle

An ultra supercritical steam cycle with double reheat forms the bottoming cycle of the 1722°C firing temperature gas turbine with the lower pressure ratio of 49.9 where the gas turbine exhaust temperature is around 650°C or 1200°F (actual exhaust temperature for this case is 656°C or 1212°F). The steam cycle consists of the following conditions:

- Supercritical HP at 376 bara / 621°C or 5455 psia / 1150°F
- 1st Reheat at 166.5 bara / 621°C or 2415 psia / 1150°F
- 2nd Reheat at 24.8 psia / 621°C or 360 psia / 1150°F
- LP Steam Induction at 3.17 bara or 46 psia

The results of the cycle analysis indicate that the net IGCC plant heat rate is essentially unaffected by installing an advanced steam bottoming cycle. Such advanced steam cycles show a significant advantage in lowering the heat rate in a boiler plant because the amount of high temperature heat is significantly higher than that in available in the exhaust of a gas turbine.

CONCLUSIONS

Based on the results of this screening study, the cycles in the order listed below are recommended for the SubTask 1.4.2, “Advanced Brayton Cycle Detailed Analysis.” Sensitivity analysis on each of these selected cycles in an integrated gasification based power plant is recommended to quantify the effect of varying the firing temperature (along with the corresponding blade metal / TBC temperatures and pressure ratio) with the ultimate goal of achieving the efficiency target of this program while minimizing the firing temperature increase over that of the Baseline case. The cycle conditions investigated during this screening analysis provide a bases and “starting points” for the next detailed study consisting of developing the performance of the integrated plants. Sensitivity to letting the gas turbine exhaust temperature rise above the 650°C or 1200°F constraint used in the Screening Study is also required. Appropriate advanced steam cycle conditions will be utilized corresponding to the higher gas turbine exhaust temperatures. Incorporation of the High Efficiency Exhaust Diffuser is also recommended as part of the sensitivity analysis.

The overall plant integration scheme for each of these promising cycles is recommended, i.e, a total systems approach to design. An example is determining the optimum generation rate of high pressure N_2 in the ASU for injection into the gas turbine which in turn establishes the optimum amount of syngas humidification.

Promising Cycles

1. Steam-cooled Simple Cycle Gas Turbine based Combined Cycle
2. Steam-cooled Reheat Gas Turbine based Combined Cycle
3. Steam-cooled Intercooled and Reheat Gas Turbine based Combined Cycle
4. Oxy Combustion Gas Turbine (GTI Technology)
5. Water-cooled HAT – Combined Cycle
6. Vortex Combustion Gas Turbine based Combined Cycle
7. Pressure Gain Combustion Gas Turbine based Combined Cycle.

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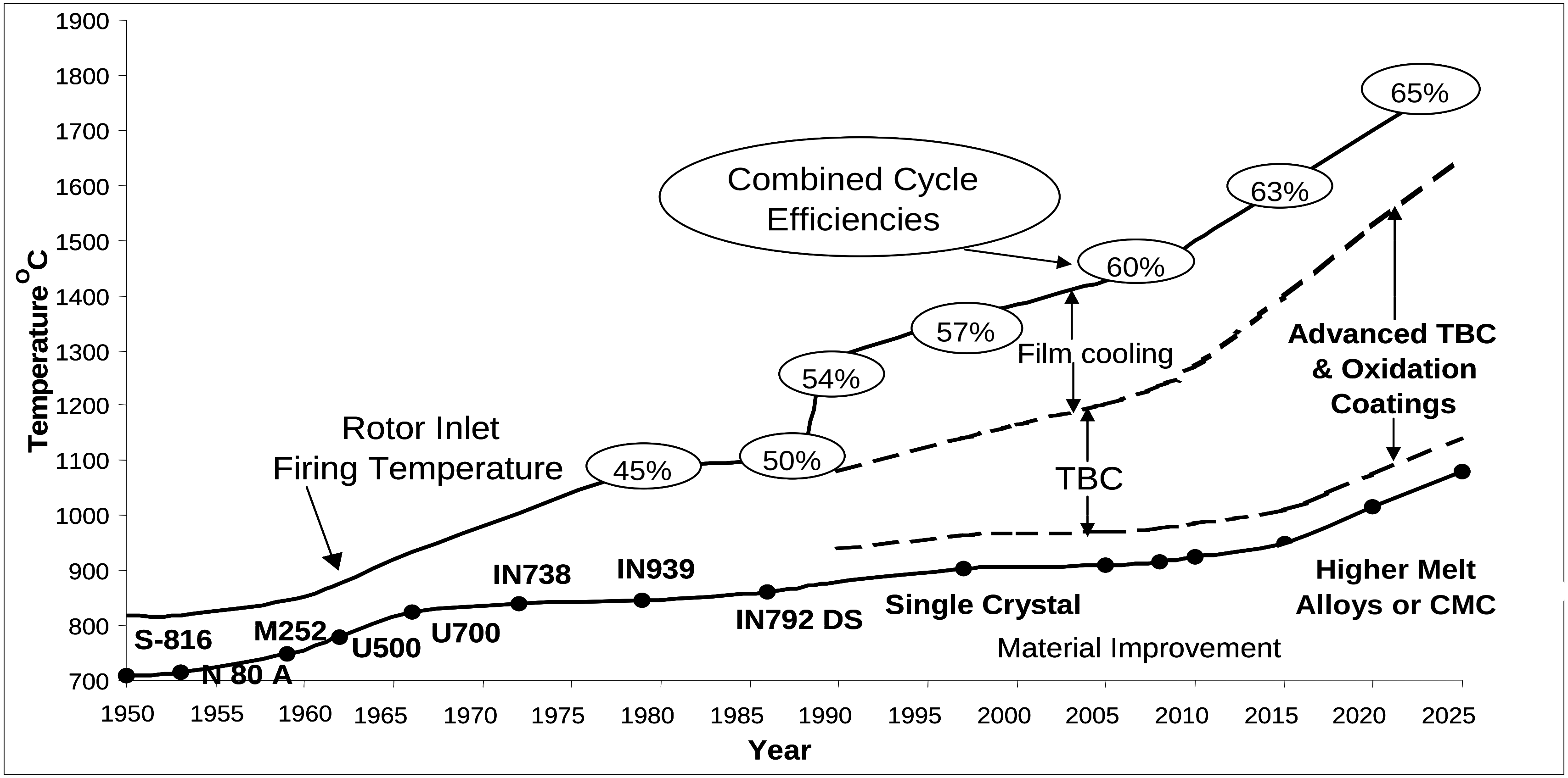


Figure A.1.4.1 - 1: Effect of Increasing Firing and 1st Stage Stator Blade Temperatures

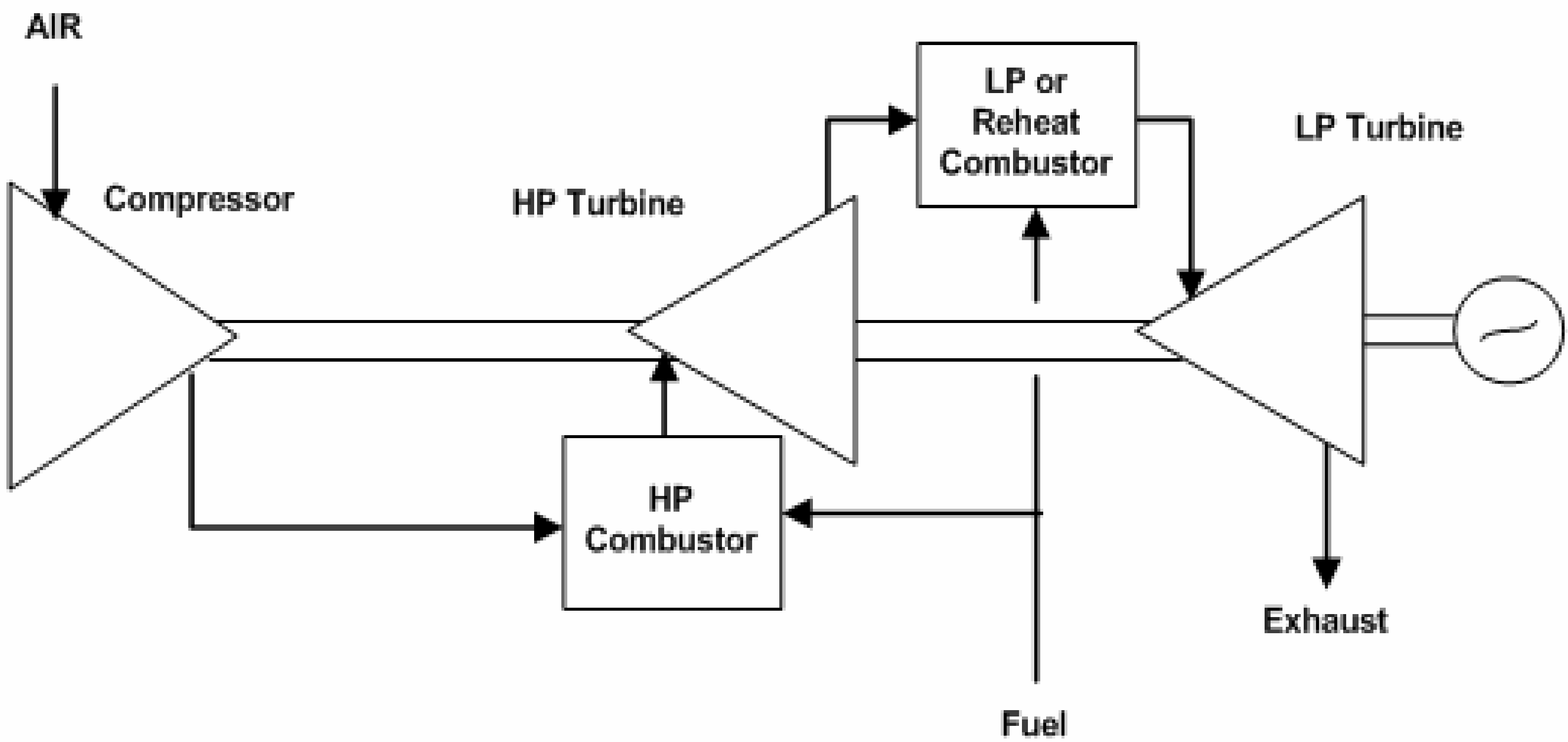


Figure A.1.4.1 - 2: Reheat Gas Turbine Cycle

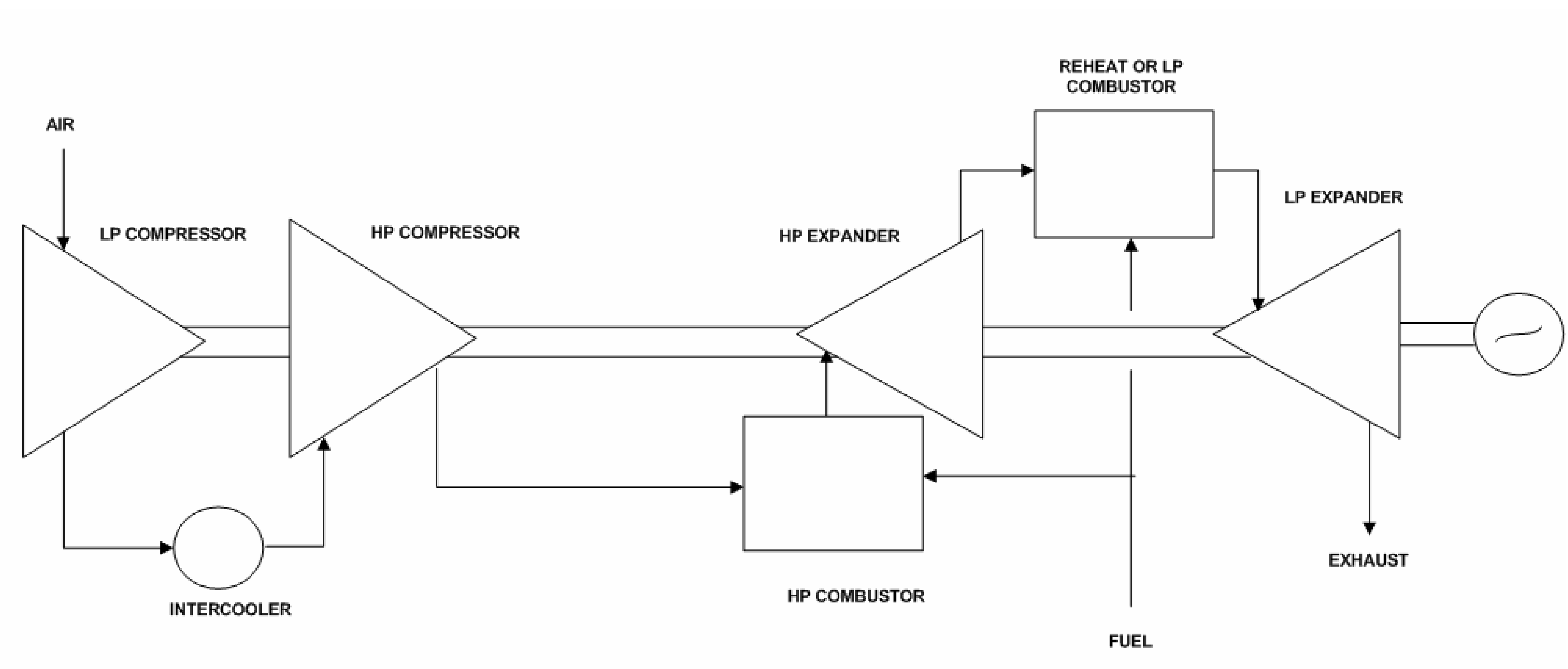
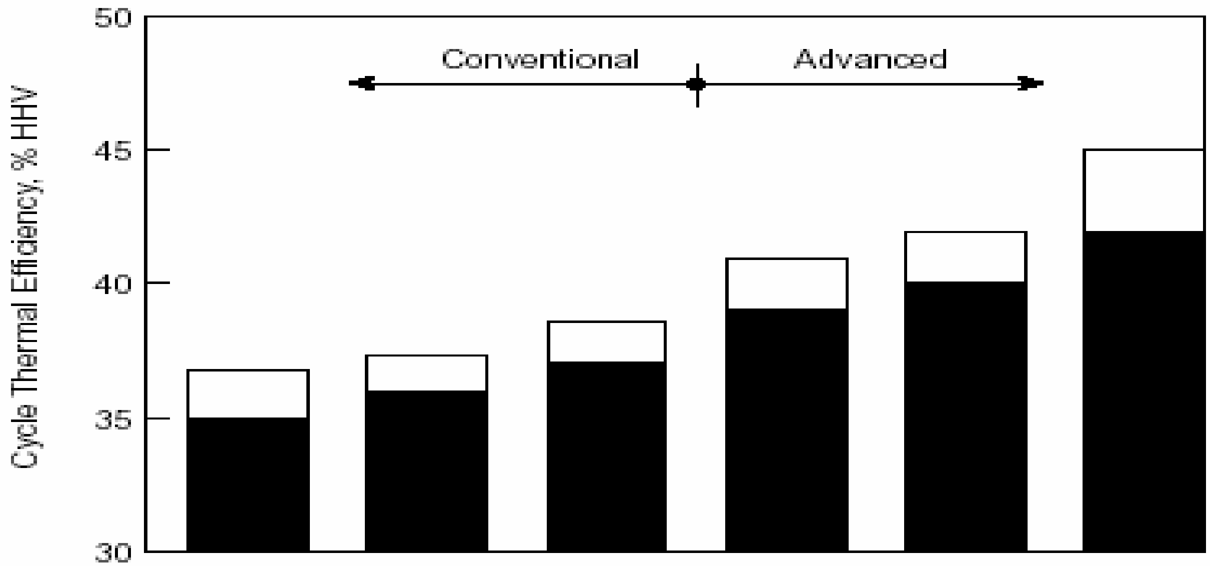


Figure A.1.4.1 - 3: Intercooled - Reheat Gas Turbine Cycle



Pressure, psi	2400	3500	3500	4500*	4500*	6000*
SH Temp, F	1000	1000	1000	1050	1100	1200
RH Temp, F	1000	1000	1025	1050	1100	1200
2RH Temp, F	—	—	1050	1050	1100	1200
Pressure, bar	16 6	24 1	24 1	31 0	31 0	41 4
SH Temp, C	538	538	538	566	593	649
RH Temp, C	538	538	552	566	593	649
2RH Temp, C	—	—	566	566	593	649

Figure A.1.4.1 - 4: Steam Rankine Cycle Thermal Efficiencies

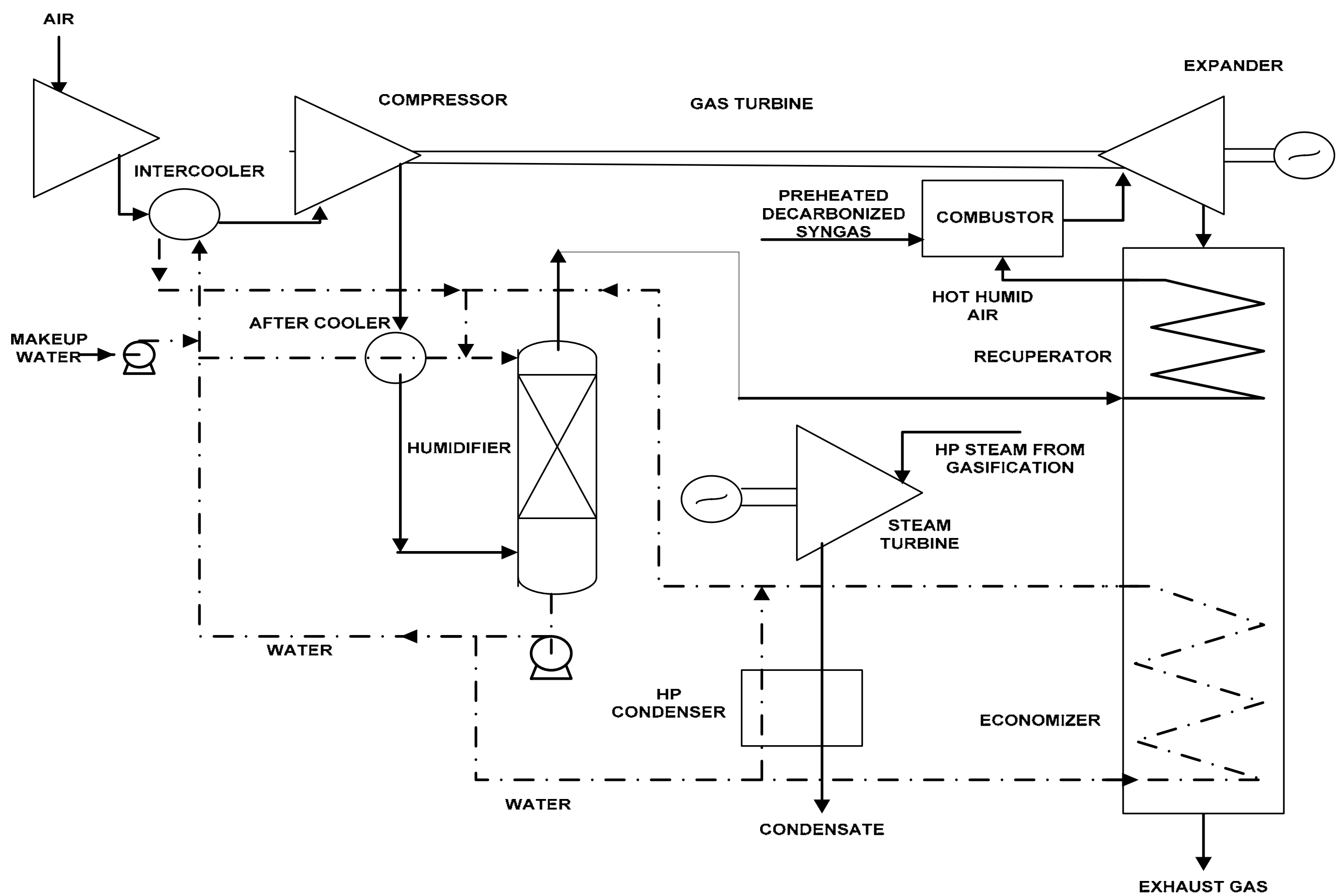


Figure A.1.4.1 - 5: HAT-Combined Cycle

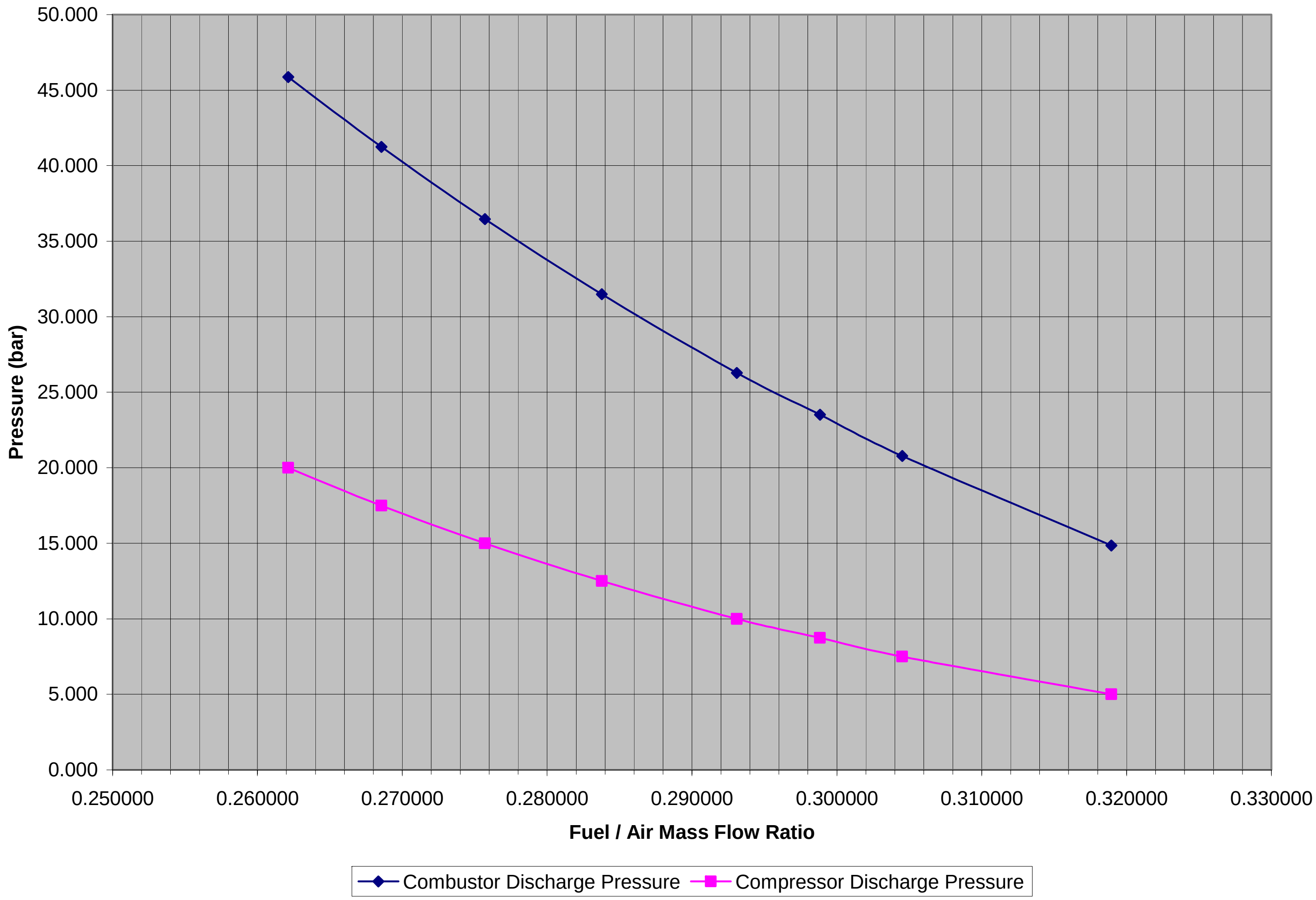


Figure A.1.4.1 - 6: Pressure Gain Combustor

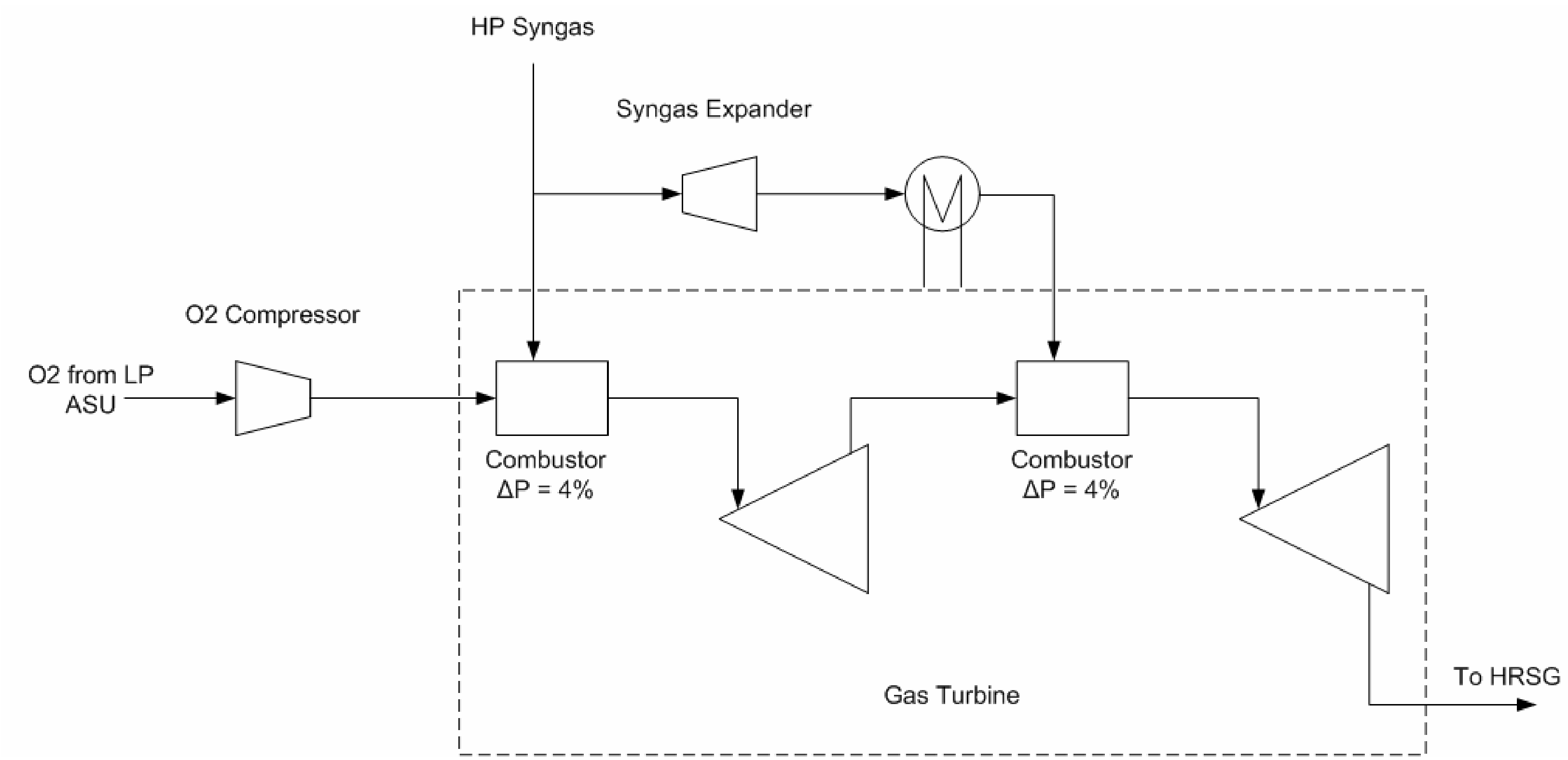


Figure A.1.4.1 - 7: OX Gas Turbine

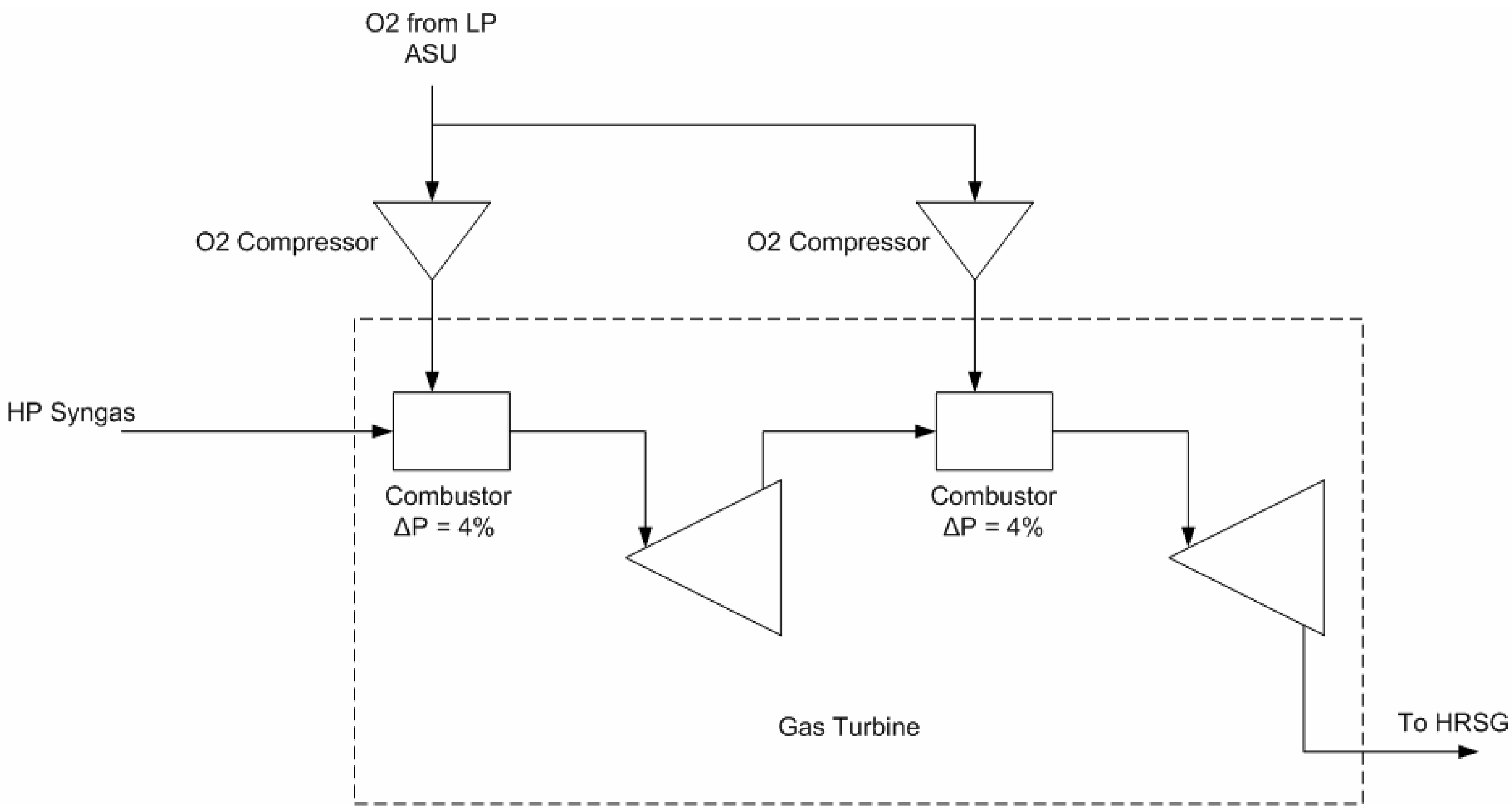


Figure A.1.4.1 - 8: POX Gas Turbine